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I, MARIO PERUSSICH, ASSISTANT DIRECTOR PATENT SERVICES, hereby certify that the annexed are true copies of the Provisional specification and drawing(s) as filed on 21 November 1996 in connection with Application No. PO 3739 for a patent by AIMBRIDGE PTY LTD filed on 21 November 1996.

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PROVISIONAL SPECIFICATION

Applicant(s):

AIMBRIDGE PTY LTD

A.C.N. 054 510 404

Invention Title:

DOUBLE ORBITAL TRANSMISSION

The invention is described in the following statement:

DOUBLE ORBITAL TRANSMISSION

This invention relates to a double orbital transmission and, in particular, to an improvement or modification to the transmission disclosed in our earlier International Patent Application No. PCT/AU94/00445 (publication No. WO-95/06829). The contents of the earlier application are incorporated into this specification by this reference.

As is disclosed in the above international application, infinitely variable transmissions which operate on a friction principle are well known. One such transmission comprises a vee-belt and pulley system. The pulleys are each split into two frusto-conical portions which are movable axially towards or away from each other so as to vary the effective pulley diameter at which the belt contacts the pulley. The major problem with this and other friction transmissions is that they are unable to transmit high torques, at least without making a transmission of excessive size as to be impractical.

A requirement accordingly exists for a variable ratio transmission which is able to transmit high torques in a practical manner. Transmission systems capable of coping with large torque loads in relatively small units, are inevitably based on rigid body elements such as gears formed in metals. This poses great problems for infinitely variable transmissions.

There is disclosed in my patent application PCT/AU81/00146 an infinitely variable mechanical transmission. Basically, this mechanism comprises means for transforming a circular input motion into non-circular periodic motion of a plurality of elements, or iterated operations of a single element, utilising only a part of the periodic motion of each element and transforming this part back into a rotary

output motion. These parts of the periodic motion of the plurality of elements are connected or "assembled" sequentially to provide the output motion. This process is what is termed "motion transformation" and results in so-called "torque conversion".

In the transmission disclosed in International Patent Application No. PCT/AU81/00146, rotary motion of an input shaft is converted by an eccentric of variable eccentricity into a periodic motion of a plurality of racks. The periodic motion of each rack is converted into a rotary periodic motion of a pinion, and a selected part of the motion of the pinion is applied to a separate satellite gear of a planetary gear arrangement. The resultant output motion of a sun gear of the planetary gear arrangement is effected by the sequential action of each satellite gear. More specifically each rack operates in sequence to apply part of its motion to is associated satellite gear and thus to the output sun gear, a switching device being incorporated in the mechanism to switch on and off an operative connection between a pinion gear which is continually driven by the rack, and the associated satellite gear. While it may be theoretically possible to achieve either instantaneous switching or precise synchronism between the switching off of the operative connection between one rack and its associated satellite gear and the switching on of the operative connection between the next rack in the sequence and its associated satellite gear, it is not possible in practice to achieve this, and as a result the output will not be completely smooth; this may manifest itself as a slight jerkiness which can be felt in the output while under load. Whereas for some uses this lack of smoothness may be tolerated, for many uses it is necessary to obtain a flat or smooth and continuous output.

Thus, attempts to produce rigid body continuous variable transmissions have been based on the production of a plurality of partial intermediate circular or non-circular motions produced by a circular input and at some stage transformed back to a collated circular motion.

Pires U.S. Patent No. 4,983,151 issued 8 January 1991 discloses a mechanism which attempts to provide a smooth output by what Pires terms "averaging intermediate rotations". The device disclosed in Pires requires considerable precision and whilst the output is smoother than the transmission referred to in PCT/AU81/00146, the output still is not sufficiently smooth for many applications.

Our earlier International Patent Application No. PCT/AU94/00445 discloses the use of load distributing means for differentially distributing the load taken by secondary members of the transmission so that the load is distributed between at least two such members at any one time. Because of this distribution of load, the output power provided by the secondary members is smoother and continuous rather than jerky and discontinuous and therefore the transmission of input power to output power is smoother than in prior art rigid body continuously variable transmissions. The load distributing means which differentially distributes the load, collapses the kinetic form of the overlapping partial circular or non-circular motions and serially links their associated load functions by differentially distributing the load between at least two of the secondary members.

The specific embodiments disclosed in International Patent Application No. PCT/AU94/00445 are directed to bicycle transmissions, winches and other generally slow moving mechanisms although the invention is applicable to any type

of transmission which requires or could use continuous variation in the drive output between a minimum drive ratio and a maximum drive ratio.

The present invention stems from further development of the invention disclosed in International Patent Application No. PCT/AU94/00445 and which, whilst could be used in any application requiring or desiring continuously variable transmission from a minimum ratio to a maximum ratio, is more concerned with higher speed and higher power applications such as heavy duty winch applications and automotive applications.

The invention may be said to reside in a transmission including:

an input means;

an output means;

a plurality of secondary members for supplying output power for only part of each rotary cycle of the input means;

power transfer means for engagement with the plurality of secondary members;

the plurality of secondary members being coupled to one of the input means or the output power supply and the power transfer means being coupled to the other of the input means or the output means;

first orbital means for causing the plurality of secondary members to undergo orbital motion; and

second orbital means for causing the power transfer means to undergo orbital motion so the combined orbital motions cause power to be transmitted from the input power supply to the output power supply.

Preferably the transmission further includes load distributing means for differentially distributing the load taken by the secondary members between at least two of the

secondary members at any one time.

Preferably the transmission includes phase changing means for changing the phase relationship of the orbital motions to, in turn, change the drive ratio of the transmissions.

Preferably the orbital motion is a stationary orbital motion but in other embodiments the orbital motion could be either a progressive or a regressive orbital motion.

Preferably the secondary members comprise a first set of pawls and a second set of pawls.

Preferably the first orbit means comprises a pawl carriage for carrying the first and second sets of pawls, the pawl carriage having an epicyclic plate, an orbital control plate adjacent the epicyclic plate and orbit control means between the orbital control plate and the epicyclic plate.

Preferably the orbit control means comprises a hole or recess on one of the orbital control plate or epicyclic plate and pins for engaging the hole or recess on the other of the orbital control plate or epicyclic plate.

In other embodiments, the orbit control means may comprise a gear recess on one of the epicyclic plate or orbit control plate and a gear member, for receipt in the gear recess, on the other of the epicyclic plate or orbit control plate; or a recessive or progressive orbital gear arrangement.

Preferably the power transfer means comprises a first assembler ring for engaging with the first set of pawls and a second assembler ring for engaging with the second set of pawls.

Preferably the first and second assembler rings have ratchet teeth on an inner peripheral surface and the pawls carry shoes which in turn have ratchet teeth for engaging with the ratchet teeth on the first and second assembler rings.

Preferably the pawl-carriage has an axial portion and the pawls are pivotally coupled to the axial portion of the carriage ring.

Preferably the second orbit means comprises an orbit body for carrying the first and second assembly rings, the orbit body having an epicyclic plate, an orbital control plate adjacent the epicyclic plate and orbit control means between the orbital control plate and the epicyclic plate.

Preferably the orbit control means comprises a hole or recess on one of the orbital control plate or epicyclic plate and pins for engaging the hole or recess on the other of the orbital control plate or epicyclic plate.

In other embodiments, the orbit control means may comprise a gear recess on one of the epicyclic plate or orbit control plate and a gear member, for receipt in the gear recess, on the other of the epicyclic plate or orbit control plate; or a recessive or progressive orbital gear arrangement.

Preferably the input means comprises a first input shaft having an eccentric upon which the pawl carriage is mounted and a second input shaft having an eccentric upon which the orbit body is mounted.

Preferably the input means also includes phase control means for controlling the phase relationship between the first and second input shafts and therefore between the

first and second eccentrics to in turn control the phase relationship between the first and second eccentrics and therefore the phase relationship between the orbital motions.

Preferably the differential load distribution means comprises differential load distribution gears arranged between the first and second assembler rings so that load can be transmitted from the first assembler ring to the second assembler ring and vice verse to thereby differentially distribute load between one of the first set of pawls and one of the second set of pawls at any one time.

Preferably the engagement shoes are guided in a guide ring arranged between the first and second assembler rings.

Preferably the engagement shoes have guide flanges which are received in grooves in the guide ring to thereby guide movement of the engagement shoes relative to the guide ring and the first and second assembler rings.

Preferably the differential load distribution gears are mounted on the guide ring and engage bevel teeth on side surfaces of the first and second assembler rings.

In this embodiment of the invention, the pawls are mounted on the pawl carriage which is in turn arranged on the first eccentric and the assembler rings are arranged radially outwardly with respect to the pawls.

In one embodiment of the invention, the teeth on the assembler rings which engage with the teeth on the engagement shoes are ratchet teeth.

In another embodiment of the invention, positive engagement

means is provided for moving the pawls into a position where the two orbits are able to positively cause engagement between the pawls and the assembler rings for any given phase relationship between the orbits. In this embodiment, the teeth on the assembler rings which are to engage the pawls are of sinusoidal shape. In this embodiment of the invention, the pawls may be provided with teeth at their ends rather than engagement shoes and preferably the teeth are also of sinusoidal shape. However, the pawls could be provided with engagement shoes having teeth of sinusoidal shape.

In this embodiment of the invention, the assembler rings are mounted on the first eccentric and the pawls are arranged radially outwardly of the assembler rings for engagement with the assembler rings.

Preferably the positive engagement means comprises arm members on the pawls and a control body for axial movement relative to the pawls, the control body having wedge-shaped recesses for receiving the arms so that upon axial movement of the control body, the wedge-shaped recesses contact the arms to move the pawl bodies radially to thereby cause positive engagement of the pawls with the assembler rings.

In this embodiment, the pawls are supported by the orbit body and the orbit body is provided with openings for receiving the pawls.

Preferably control means is provided for axially moving the control body to engage and disengage the pawls with respect to the assembler rings.

In a further embodiment of the invention, the first orbital means includes a first eccentric and orbit control means for controlling the orbital motion and the second orbital

means comprises a plurality of axles from which is mounted the power transfer means, the axles having eccentrics and being rotatable to provide controlled orbital motion to cause the power transfer means to undergo orbital motion.

Preferably the power transfer means are supported by an orbital body mounted on the axles.

Preferred embodiments of the invention will be disclosed, by way of example, with reference to the accompanying drawings, in which:

Figure 1 is a partially broken-away view of a transmission embodying the present invention;

Figure 2 is a cross-sectional view of the transmission of Figure 1;

Figure 3 and Figure 4 are diagrams illustrating the principle of the invention in schematic form;

Figure 5 is a view of an alternative arrangement which can be used in the embodiment of Figures 1 to 4;

Figure 6 is a partially cut away view of a second embodiment of the invention;

Figure 7 is a cross-sectional view of the embodiment of Figure 6;

Figure 8 is a view along the line VIII-VIII of Figure 7;

Figure 9 is an illustrative diagram used to explain operation of the embodiment of Figures 6 to 8;

Figures 10A and 10B are operational diagrams relating to the operation of the embodiment of Figures 6 to 8;

Figures 11A and 11B are operational diagrams similar to Figures 10A and 10B;

Figures 12 to 14 are operational diagrams relating to yet a further embodiment of the invention; and

Figure 15 is a view of a further embodiment of the invention.

With reference to Figures 1 and 2, a transmission 10 is shown which has a first input shaft 12 which carries a first eccentric 14. The input shaft 12 is hollow and arranged within it is a second input shaft 13 which carries a second eccentric 16. A pawl carriage 20 is arranged on the eccentric 13 via bearings 22. The pawl carriage 20 has an axially extending portion 24 and a radially extending portion 26. The axially extending portion 24 has two circumferential grooves 28 and 30 in which two sets of pawls 32 and 33 are respectively pivotally mounted via pivot pins 34. It should be noted in the cross-sectional position shown in Figure 2 the pawls 33 associated with the grooves 28 cannot be seen because they are out of alignment with the pawls 32 arranged within the groove 30 as is most clearly shown in Figure 1. The number of pawls in each set of pawls 32 and 33 which can be pivotally mounted in each of the grooves 28 and 30 is arbitrary but preferably comprises from four to eight pawls in each groove 28 and 30.

The pawls 32 and 33 are provided with engagement shoes 36 and 37 respectively. The engagement shoes 36 and 37 are pivotally mounted to the pawls 32 and 33 by pivot pins 38. The engagement shoes are provided with ratchet teeth 40 as is best seen in Figure 1.

First and second assembler rings 42 and 44 are provided about the axial section 24 of the pawl carriage 20 and are radially aligned with the respective pawls 32 and 33.

The inner circumference of the assembler rings 42 and 44 are provided with ratchet teeth 48 for engagement with the ratchet teeth 40 on the engagement shoes 36 and 37 as will be disclosed in more detail hereinafter.

An outer orbit body 50 has a first enlarged diameter

section 52 and a second smaller diameter section 54. The enlarged diameter section 52 surrounds and supports the assembler rings 42 and 44. Roller bearings 54 are provided between the assembler rings 42 and 44 and the portion 52 of the orbit body 50. The roller bearings 54 may be located in grooves 56 provided in the internal surface of the portion 52 and act to support and guide relative rotation of the assembler rings 44 and 46 relative to the orbit body 50.

A guide ring 60 is arranged between the assembler rings 42 and 44 and is fixed to the orbit body 50. The guide ring 60 can be fixed to the orbit body 50 by bolts or other suitable fasteners or alternatively could be made integral with the orbit body 50 and project radially inwardly from the orbit body 50 between the assembler rings 42 and 44. The guide ring 60 has circumferential grooves 62 and 64 on side surfaces 66 and a plurality of cutouts 68 on outer surface 70.

The assembler rings 44 and 46 have gear teeth 72 on inwardly facing surfaces 74 of the assembler rings 44 and 46. Arranged within each of the cutouts 68 is a gear 80 which is mounted on an axle 82 secured in a respective cutout 68 and to the ring 60. The gear 80 meshes with the teeth 72 on the side surfaces of the assembler rings 44 and 46 as can be best seen in Figure 1. Once again, the number of cutouts 68 and gears 80 is somewhat arbitrary but typically between four and six such gears may be provided.

Engagement shoes 36 and 37 are provided with projecting flanges 88 and 89 which engage within the grooves 62 and 64 of the ring 60 to thereby guide movement of the shoes 36 and 37 about a predetermined path relative to both the rings 60 and the assembler rings 42 and 44.

The radial portion 26 of the pawl carriage 20 forms a first epicyclic plate and carries a plurality of pins 100 which are received in circular holes or recesses 102 in a first orbit control plate 104. The plate 104 is mounted on first input shaft 12 by bearings 106. The pins 100 and recesses 102 form a first orbit control as will be described in more detail hereinafter.

The reduced diameter portion 54 of the orbit body 50 is mounted on the second eccentric 16 via bearings 108. The second portion 54 has a stepped portion 110 which forms a second epicyclic plate and which carries pins 112. The second eccentric 16 has an extension shaft 114 on which an output shaft 120 is mounted. The output shaft 120 has a radially extending flange 122 which is provided with a plurality of holes or circular recesses 124. The pins 110 are received within the holes 124 and the pins 110 and holes 124 form a second orbit control as will be explained in more detail hereinafter.

Figure 2 shows a stand or outer casing 130 which is mounted on output shaft 112 via bearings 132. The casing 130 is not shown in Figure 1 for ease of illustration and clarity purposes.

In order for the transmission 10 to provide power from the input to the output, rotary input power from a power source (not shown) is provided to the two input shafts 12 and 13. The rotary input power is supplied via a phase controller as described in our copending Australian Patent Application Nos. PN8244, PO0627 and PO1567, the contents of which are incorporated by this reference. The purpose of the phase controller is to also provide a desired phase relationship between the input shafts 12 and 14 and therefore between the eccentrics 14 and 16 to control the drive ratio of the transmission as will be described hereinafter.

Thus, input rotary power is supplied to the input shafts 12 and 13 from a power source such as internal combustion engine, electric motor or any other drive source so that the shafts 12 and 13 are rotated. Rotation of the shaft 12 will cause the eccentric 14 to rotate to in turn move the pawl carriage 20. Since the pawl carriage 20 is coupled to orbit control plate 104 (which may simply be part of the outer casing 130 or the like) the carriage 20 is restrained to undergo all stationary orbital motion because of the engagement of the pins 100 within the recesses 102 of stationary control plate 104. Similarly, rotation of the input shaft 13 rotates the eccentric 16 so that the orbit body 50 also undergoes stationary orbital motion due to the interconnection of the orbit body 54 to the flange portion 122 which forms a control plate by virtue of the engagement of the pins 110 in recesses 124. Thus, a double orbit stationary orbital motion is created. If the orbital motions are out of phase, as will be explained in more detail with reference to Figures 3 and 4, the pawls 32 and 33 will come into engagement with the assembly rings 42 and 44 (via the engagement shoes 36 and 37). Each pawl 32 and 33 in the sets of pawls will therefore engage with the respective rings 42 and 44 in turn for part of the rotary cycle of the transmission so as to transmit drive so that the rings 42 and 44 will also tend to orbit and transmit drive through to the guide ring 60 via the differential load distribution gears 80. Thus, the guide ring 60 and therefore the orbital body 50 will rotate as well as undergo stationary orbital motion. Thus, the motion of the orbital body is a complex motion involving both a stationary orbital motion as well as a rotary motion. The orbital control formed by the pins 110 and the recesses 124 acts as a transformer to separate the complex orbital and rotational motions so that the orbital motion remains with the orbital body 50 and the rotary motion is transferred to the output shaft 120 so that the output shaft is rotated to

thereby provide output rotary power.

As is best shown with reference to Figures 3 and 4, if the input shafts 12 and 13 are adjusted by the phase control mechanism of the type disclosed in the above mentioned Australian Patent Applications so that the eccentrics 14 and 16 are in phase as is shown in Figure 3, if there is no difference between the aphelion and parhelion of the two orbits ie the aphelion equals the parhelion. If the shafts 12 and 13 are adjusted by the phase control mechanism to bring about a phase difference between the eccentrics 14 and 16 as is shown in Figure 4, the aphelion and parhelion of each individual orbit remains the same but the relationship between the two changes throughout the cycle of rotation supplied to the input shafts 12 and 13 and it is this change which causes the operation of the pawls during the orbiting motion which creates the rotation of the assembler rings 42 and 44 and also the rotation of the orbit body 50 as well as the orbital motion of the orbit body 50 to thereby provide the output power supply. The ratio of the output compared to the input is adjusted by the amount of phase difference between the eccentrics 14 and 16 which, again, is controlled by the phase control mechanism mentioned above. Figure 4 shows the maximum phase difference of 180°. If the phases of the eccentrics 14 and 16 is adjusted so that there is no phase difference as shown in Figure 3, then no output power is produced.

The different phase relationship of the two orbits produces different rates of acceleration of the approach of the inner surface of the orbit body 50 and the outer surface of the pawl carriage 20 and this allows the production of different ratios of drive.

Preferably the orbital motion of the carriage 20 and orbit body 50 is a stationary orbit. However, either a

progressive or regressive orbit could also be embodied in the invention. If a progressive gear is used to create a progressive orbit, instead of an epicyclic plate (of the type shown in the drawings) or a stationary gear, then either the ratio can be increased or reverse function can be made to operate when the two orbits are in phase.

The gears 80 form differential load distribution gears which ensure that load is distributed between at least two of the pawls 32 and 33 at any one time. The differential load distribution occurs because when one of the pawls 32 comes into engagement with the ring 40, via the engagement shoe 36, the assembler ring 42 will accelerate from the beginning of the drive up to a maximum level and then begin to slow. At this point, the acceleration of the next pawl 32 which begins to come into engagement with the ring 42 becomes greater and it overtakes the first pawl 32 causing the pawl 32 to become disengaged from the motion of the system.

Similarly, when pawls 33 come into engagement with assembler ring 44 via the shoes 37, exactly the same type of acceleration occurs. The speeding up and slowing down of the rings 44 causes the differential load distribution gears 80 to rotate back and forth in a rocking type motion as the assembler rings 42 and 44 speed up and slow down relative to each other. This rocking motion will transfer the load from one of the pawls 32 to at least one of the pawls 33 to thereby provide a smooth output at the output shaft 120. The load distribution gears 80 act to provide as much differential load transfer as is required to provided a completely smooth output. The differential load distribution gears 80 thereby provide an equal distribution of the load between the pawls 32 and 33 and therefore zero fluctuation at the output shaft 120. Thus, a continuous and smooth output is achieved.

In order to change the drive ratio of the transmission, the phase relationship between the eccentrics 14 and 16 is altered by the phase adjustment mechanism according to the earlier patent applications mentioned above. Thus, the phase relationship can be changed between zero phase difference to a 180° phase difference to thereby change the drive ratio of the transmission in a continuously variable fashion. Thus, continuously variable transmission from a minimum drive ratio to a maximum drive ratio can be obtained by the transmission (for example, from a very low ratio up to a ratio approaching 1:1).

In a further embodiment of the invention, rather than provide orbit control via the pins 100 and 110, and recesses 102 and 124, recessed gears of the type shown in Figure 5 could be utilised.

In this embodiment, the holes 124 in radial portion 122 and the holes 102 in plate 104 would be replaced by a cutout 140 and the pins 100 and 110 would be replaced by a gear 142 which locates within the cutout 140 and engages in the recess 140 to control the orbital motion in exactly the same way as the pins 100 and 112 engage in the openings 102 and 124 to control the orbital motion.

The pins 100 would basically ride in approximately one half of the inner periphery of the holes or recesses 102 during the orbital motion of the carriage 20 to thereby guide the orbital movement.

With reference to Figures 6 to 8, a second embodiment of the invention is disclosed. In this embodiment the assembler rings and pawls are reversed so that the assembler rings are radially inwardly of the pawls and the pawls are radially outwardly of the assembler rings rather than the opposite arrangement which was described with

reference to Figures 1 to 5. Furthermore, this embodiment provides for positive engagement of the pawls with the assembler rings as will be described in more detail hereinafter. Like reference numerals in this embodiment relate to the same parts as described with reference to the previous embodiments.

In the embodiments of Figures 6 and 7, the pawl carriage 20 now effectively forms a carriage for the assembler rings 42 and 44 so that the assembler rings 42 and 44 are mounted on the carriage 20 which is arranged on the eccentric 14 in the same manner as previously described. The carriage 20 has a radially inwardly projecting flange 26. In this embodiment, rather than use an epicyclic plate to control the orbit, a recessive gear arrangement is utilised. In this regard, the outer circumference of the flange 26 is provided with gear teeth 107. A gear ring 109 is arranged on the bearings 106 and the gear ring 109 has an axially extending flange 111. The inner circumference of the flange 111 carries gear teeth 113 which are intended to engage with the gear teeth 107 to control orbital motion in the form of a regressive orbital gear arrangement.

In this embodiment, the differential load distribution gears 86 are carried by a ring 60 arranged between the assembler rings 42 and 44 as in the earlier embodiments and engage with teeth on the rings 42 and 44 exactly as described in the earlier embodiment.

The orbital body 50 in this embodiment has a pawl retaining section 150 which has bores or slots 152 for receiving the pawls 32 and 33. The pawls 32 and 33 have arms 156 extending radially outward thereof.

Arranged about the orbital body 50 is a cylindrical adjustment control 160. The adjustment control 160 has

wedge shaped grooves 162 and 164 which receive the arms 156 of the pawls 32 and 33. The pawls 32 and 33 do not have engagement shoes as in the earlier embodiment, but rather the teeth 40 are provided directly on the ends of the pawls 32 and 33. However, in this embodiment of the invention, rather than the teeth being ratchet type teeth, the teeth are preferably sinusoidal in shape. Similarly, in this embodiment, the teeth on the assembler rings 42 and 44 which engage with the teeth 40 are arranged on the outer peripheral surface of the rings 42 and 44 and are also sinusoidal in shape to match the teeth on the engagement shoes in such a way as sot minimise radial forces.

The adjustment control 160 has a flange 162 at one end which is engaged by a bifurcated control head 164 which is provided on a control cylinder 166. The control cylinder 166 projects through an annular space 167 in the stand or casing 130. Screw-threaded shafts 170 are engaged with screw-threaded holes 172 in blocks 168 which are provided about the circumference of the adjustment control cylinder 166. The screw-threaded shafts 170 is axially retained in a bore 176 in the casing 130. A crank handle 178 or a gear may be provided on the rods 170 for facilitating adjustment of the rods 170 as will be described hereinafter.

In order to effect positive engagement of the pawls 32 and 33 with the assembler rings 42 and 44, the handle 178 can be cranked so as to cause the adjustment cylinder 166 to move in one of the directions shown by double-headed arrow A by virtue of engagement of the screw-threads on shaft 170 with the screw-threaded bores 172 in blocks 168. Movement of the adjustment cylinder 176 also moves the adjustment control ring 160 in one of the directions of double headed arrow A by virtue of engagement of the bifurcated head 164 with the flange 152. Movement of the control ring 160 will therefore cause the wedge-shaped recesses 162 to move

relative to the arms 156 which, assuming that the adjustment control ring is moved to the left in Figure 7 will cause the arms 156 to be forced inwardly in the direction of arrow B so that the pawls 32 and 33 are moved in the same direction to push the pawls 32 and 33 into a position for a given ratio (or phase relationship) where positive engagement with the sinusoidal teeth on the rings 42 and 44 as the shoes and rings 42 and 44 are brought into engagement by the rotational and orbital movement of the carriage 20 and therefore the rings 42 and 44 and also the orbital movement of the orbiting body 50 which carries the pawls 32 and 33. Thus, the two out of phase orbital motions therefore cause power to be transmitted from the input shafts 12 and 13 from the assembler rings 42 and 44 to the pawls 32 and 33 and hence to the orbiting body 50 to cause rotation as well as orbital motion of the orbiting body 50 which in turn is supplied to the output shaft 120 via the flange 12 which forms an epicyclic plate for orbital control of the orbiting body 50.

Figures 9, 10A, 10B, 11A and 11B are illustrative diagrams showing operation of the embodiments of Figures 5 to 8. Figure 9 is a diagram showing the assembler rings 42 and 44, the differential load distribution gear 62 between those rings 42 and 44 and the pawls 32 and 33. Obviously, only one of the pawls 32 and one of the pawls 33 are shown for illustrative purposes. Figures 10A and 10B and Figures 11A and 11B show the assembler ring 42 and associated pawl 32 in the assembler ring 44 and associated pawl 33 side by side so that the relationship between the rings 42 and 44 and pawls 32 and 33 can be more easily shown and explained. Figure 10A shows that the pawl 32 is engaged with the ring 44 and that the pawl 33 is not engaged with ring 44 because of the orbital position of the assembler rings 42 and 44 with respect to the orbital body 50. Figures 11A and 11B show the orbit moved 90° in a clockwise direction. Pawl 33

has arrived at its engagement position not properly aligned with the teeth in the assembler ring 44. In order for the pawl 33 to properly engage now it must force (let us assume) assembler ring 44 to rotate (relative to assembler ring 42) backwards in the direction of arrow C. It should be noted that rotation in the opposite direction may also occur for proper engagement to take place since a chaotic situation governs the actual point of engagement. The operational principles would be the same regardless of in which direction rotation actually takes place. Now pawl 32 should have disengaged from assembler ring 42 but the relative anticlockwise rotation of assembler ring 44 has caused a similar clockwise rotation of assembler ring 42 because of the differential load distribution gear 80 arranged between the assembler rings 42 and 44. This has therefore forced a temporary re-engagement of pawl 32 with assembler ring 42 at the point X shown in Figure 11A. All of this has occurred without affecting the overall clockwise movement of the assembler rings 42 and 44 caused by the engagement between the rings 42 and 44 by the gears 80. The differential load distribution gears 80 have been used for the subsidiary function of achieving apparently smooth engagement without affecting the overall movement of the transmission. Thus, the differential load distribution gears 80 in the arrangement shown in Figures 6 to 8 as explained with reference to Figures 9, 10A, 10B, 11A and 11B provides two degrees of freedom of the differential load distribution mechanism provided by the gears 80. The two degrees of freedom provide the operation of the differential load distribution mechanism (namely the gears 80) to both allow for smooth engagement of the pawls 32 and 33 with the assembler rings 42 and 44 and also the differential load distribution which is required to collapse the partial motion torque functions and allow them to be assembled in a smooth continuous manner as is more fully explained in earlier International Patent Application

No. PCT/AU94/00445. Thus, the differential load distribution mechanism in this embodiment provides a further advanced function of not only flattening out the output curve to provide for smooth and continuous output power, but also allows for proper engagement of the sinusoidal teeth on the engagement shoes of the pawls with the sinusoidal teeth of the assembler rings 42 and 44.

The operation of the embodiments of Figures 6 and 7 is generally the same as the embodiment of Figures 1 and 2. The double orbital motion which is created by the eccentrics 14 and 16 and the orbital control produces a complex orbital and rotary motion from the assembler rings 42 and 44 which are mounted on the eccentric 14 and the pawls 32 and 33 which are coupled to the orbit body 50 so that the orbit body undergoes the complex rotary and orbital motion. Once again, the orbital control provided by the pins 110 and recesses 124 acts to transform the complex motion so that the orbital motion remains with the orbital body 50 and the rotary motion is supplied to the output shaft 120 so that output power is supplied to the shaft 120.

In this embodiment of the invention, rather than using stationary orbital motion at the carriage 20, a regressive orbital gear system is utilised. The regressive orbital gear system enables the output drive ratio to go through to reverse gear. In this regard, the transmission effectively stops providing output power at the output shaft 120 at a point before the eccentrics 16 and 14 are brought into phase with one another. As the phase relationship passes that point towards phase match between the eccentrics 16 and 14, the regressive gear system 107 and 113 places the transmission into reverse gear so reveres function can take place.

If the regressive gear arrangement referred to above is changed to a progressive gear arrangement by making the gear 111 orbit about the gear 107, then an increase in drive ratio can be obtained. A further increase can be obtained by making the orbital control between the orbital body 50 and flange portion 112 into a progressive gear system rather than a stationary orbiting system as disclosed.

Once again, in this embodiment of the invention, in order to change the drive ratio of the transmission, the phase relationship between the shafts 12 and 13 and therefore the eccentrics 14 and 16 is altered by the phase adjuster mechanism disclosed in the above mentioned Australian patent applications. The radial position of the pawls 32 and 33 need also be adjusted depending on the drive ration which is selected and therefore the position of the control 160 needs to be adjusted as the phase difference between the shafts 12 and 13 is adjusted. This can also be achieved by the phase control mechanism of the aforesaid patents by simply making the phase control mechanism control three shafts, namely the input 12, the input 13 and the control rod 176 instead of just two shafts. Thus, the phase adjustment mechanism can automatically change the phase relationship between the eccentrics 14 and 16 and also cause the rod 176 to rotate to thereby shift the control 160 dependent on the phase relationship between the shafts 12 and 13 to thereby position the pawls 32 and 33 in the correct position for the particular drive ratio to correctly engage with the assembler rings 42 and 44.

Although this embodiment of the invention preferably uses sinusoidal shaped teeth, it is possible other geometric shapes could be used. It is preferred not to use ratchet shaped teeth of the embodiments of Figures 1 to 5 because of the difficulty of engagement and sinusoidal or like

shaped teeth provide for slippage movement of the gears over one another to effect the counter rotation referred to with reference to Figures 11a and 11b which is required to provide good engagement of the pawls 32 and 33 with the rings 42 and 44 which minimises radial forces.

Figures 12 to 14 show a further embodiment of the invention in which the pawls 32 and 33 are provided with engagement shoes 36 and 37 of the type described with reference to Figures 1 to 5. In this embodiment of the invention, only one degree of freedom is provided for and in this embodiment of the invention, the engagement shoes 36 and 37 may be guided in guide grooves similar to that described with reference to Figures 1 to 5 rather than by the type of positive engagement mechanism shown with reference to Figures 6 to 9 and sliding movement in the orbiting body. The fact that only one degree of freedom is available in this embodiment means that some kind of backlash or looseness in engagement may be required in order to release sufficient differential load distribution to carry out the primary function of the differential load distribution mechanism in smoothing out the output power supply obtained by the mechanism.

Figure 15 shows a further embodiment of the invention. For ease of illustration some of the componentry has been omitted so as to merely represent the difference between the embodiment of Figure 15 and the embodiment of Figure 1.

In this embodiment a first input shaft 12 has a first eccentric 14 and a second input shaft 13 surrounds the first input shaft 12. The eccentric 14 has a pawl carriage 20 which has grooves 56 in which the pawls (not shown) identical to pawls 32 and 33 are located. An output shaft 120 includes a flange portion 122 which forms a control plate and the carriage 20 has a flange 26 which forms an

epicyclic plate. As in earlier embodiments, pins 100 and recesses 102 are provided for controlling the orbital motion of the carriage 20. The casing or stand 130 includes an annular support section 131 and an end plate 133. Arranged between the annular section 131 of the end plate 133 are a plurality of axles 180, such as four axles (only two of which are shown in Figure 15). The axles 180 carry eccentrics 182 which are arranged with the annular section 131 and end wall 133.

Orbit body 50 is mounted on the axles 180 and the orbit body 50 supports assembler rings 42 and 44 as in the embodiment of Figure 1. A guide ring 60 is also fixed to the orbit body 50 and the guide ring 60 carries differential load distribution gears 80 as also described with reference to Figure 1. The assembler rings 42 and 44 may be guided in the guide rings as also described with reference to Figure 1.

The axles 180 are provided with sprockets 186 and a chain 188 is arranged about the sprockets 186. The input shaft 13 is also provided with a sprocket 190 which receives the chain 188.

When input power is supplied to the input shafts 12 and 13 in precisely the same manner as described with reference to Figure 1, the eccentric 14 undergoes controlled orbital motion in view of the fixing of the carriage 20 to the control plate 122 by the pins 110 and recesses 124. The orbit body 50 and the assembler rings 42 and 44 also undergo orbital motion. However, instead of the orbit being produced by a second eccentric as in the embodiment of Figure 1, the orbit is produced by the axles 180 and the eccentrics 182 so that the two orbital motions cause transmission of power from the input shafts 12 and 13 to the output shaft 120 as in the earlier embodiment except

that in this embodiment, the carriage 20 and the pins 100 and recesses 102 which control the orbit of the carriage 20 basically provide the transformer for transforming the complex rotary and orbital motion of the carriage 20 so that the orbital motion remains with the carriage 20 and the rotary motion is supplied to the output shaft 120 to provide output drive.

In this embodiment of the invention, the orbital motion of the orbit body 50 and therefore the assembler rings 42 and 44 is automatically controlled by virtue of the axles 180 and the eccentrics 182 without the need to provide orbital control in the form of the epicyclic plate 26 and associated pins and recesses 100 and 102 or the use of regressive or stationary orbital gears.

The arrangement in Figure 15 provides a more compact transmission in terms of the axial length of the transmission and locates the pawls and assembler rings 42 and 44 more centrally as compared to the embodiment in Figures 1 and 2 where the pawls 32 and 33 and assembler rings 42 and 44 are offset to one side of the transmission. The arrangement in Figure 15 thereby overcomes problems with torque loading in the transmission and the need for heavy bearings such as the bearings 132 and 108 shown in Figures 1 and 2 which are required because of the torque loading in the embodiments of Figures 1 and 2.

Instead of using a chain 188 to transmit power from the shaft 13 to the axles 180, solid gearing may be used.

In the preferred embodiments described particularly with reference to Figures 3 and 4. The eccentrics 14 and 16 can be brought into phase relationship so that the two orbits match. It will be possible to design the eccentrics so that it is never possible to make the two orbital motions

match thereby preventing the transmission from producing drive from the input to the output in the embodiment of Figure 1.

Since modifications within the spirit and scope of the invention may readily be effected by persons skilled within the art, it is to be understood that this invention is not limited to the particular embodiments described by way of example hereinabove.

Dated this 21st day of November 1996

AIMBRIDGE PTY LTD
By Its Patent Attorneys:

GRIFFITH HACK
Fellows Institute of Patent
Attorneys of Australia.

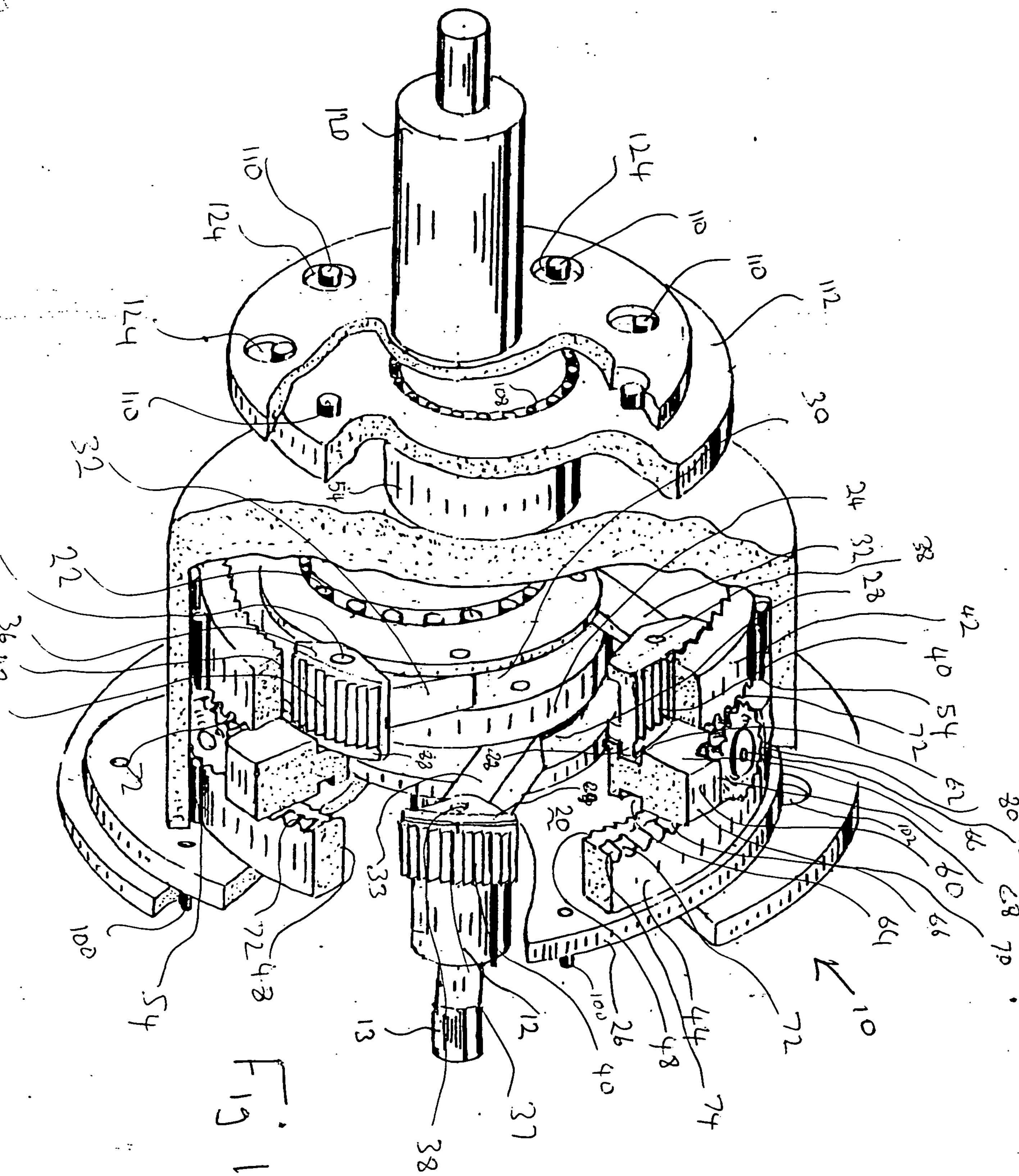


Fig 1

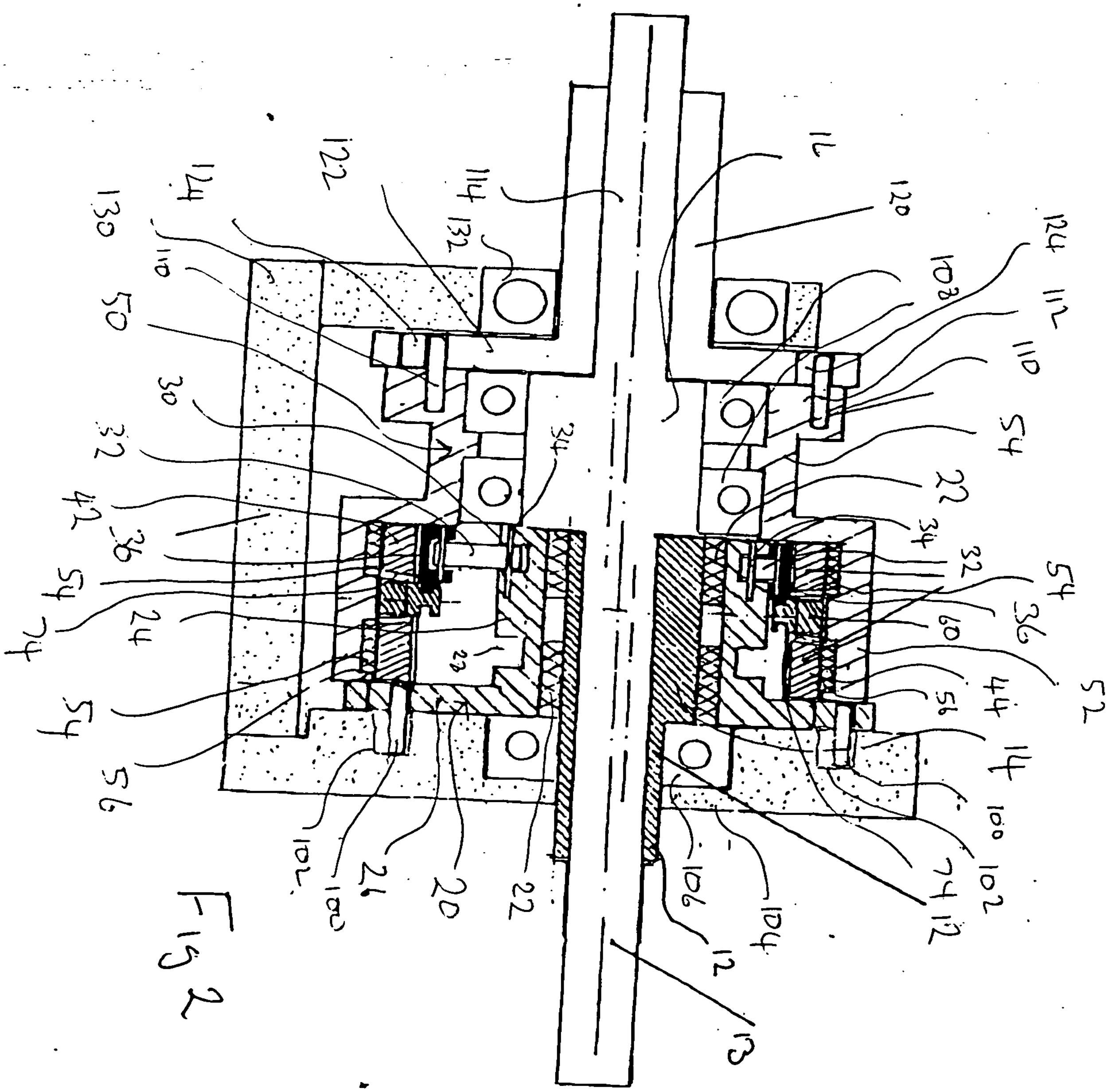


Fig 2

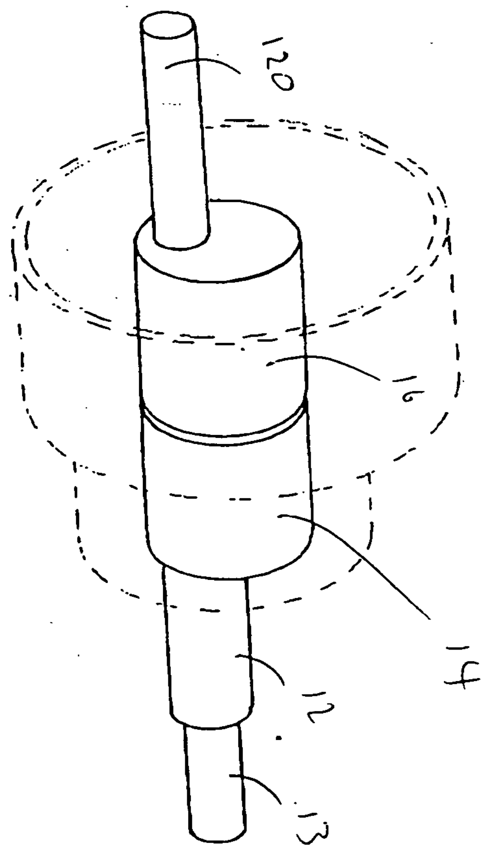


Fig 3

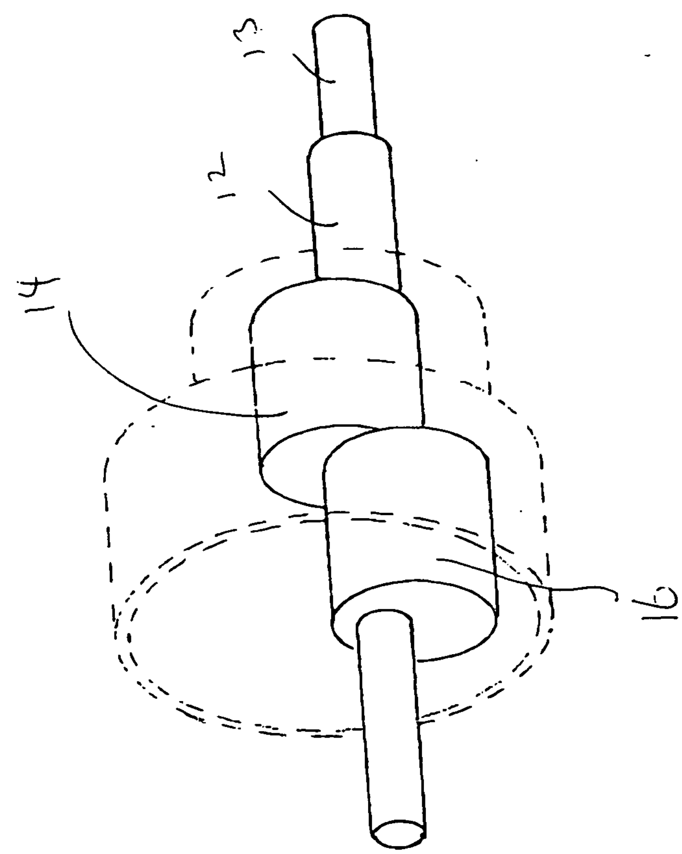


Fig 4

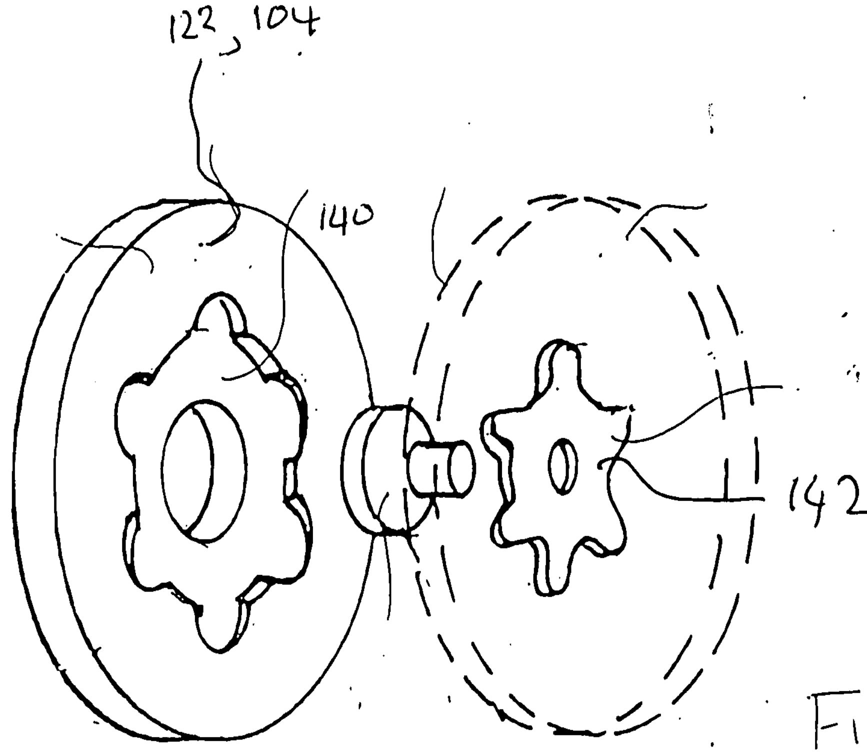


Fig 5

SHEET 4

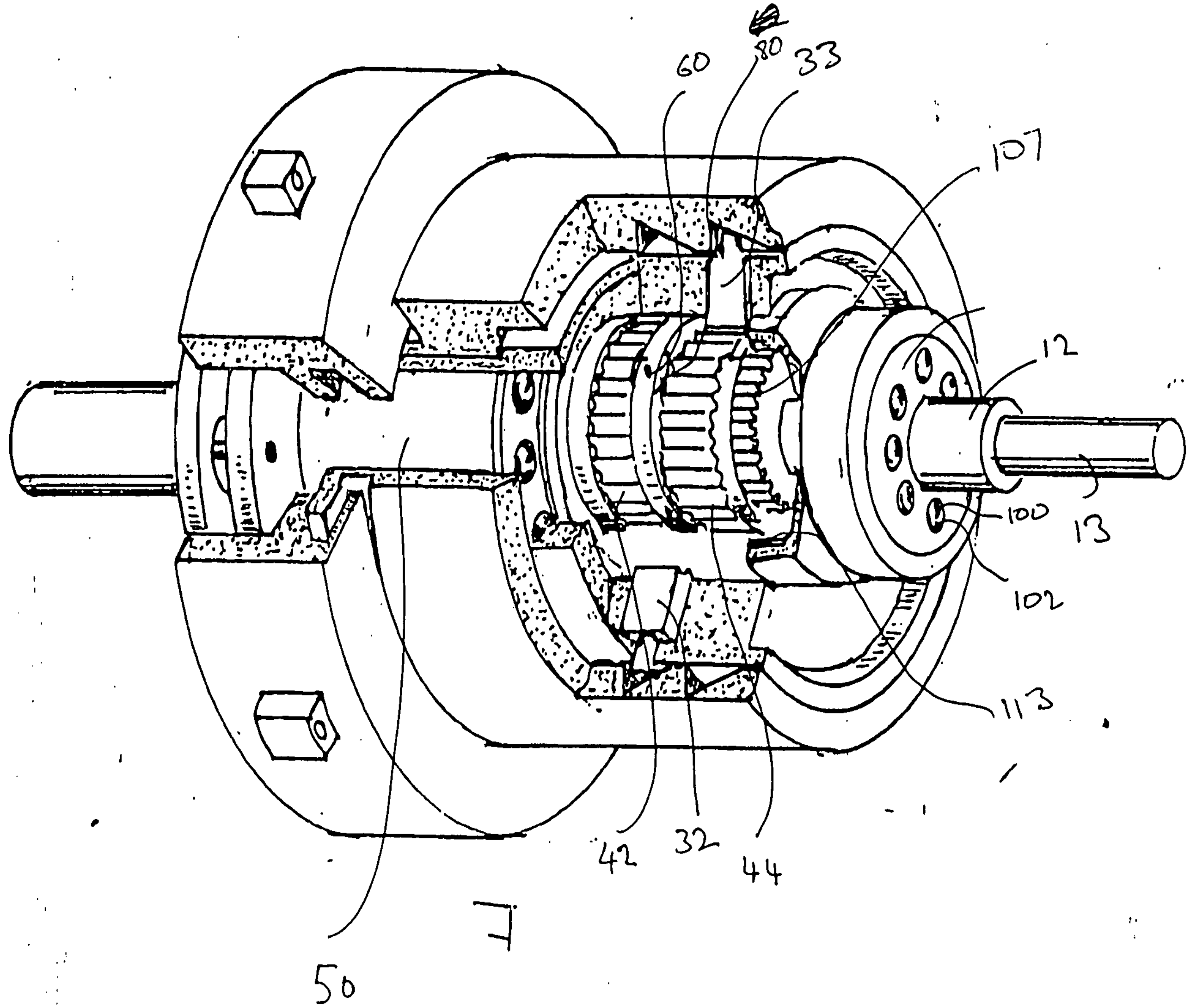
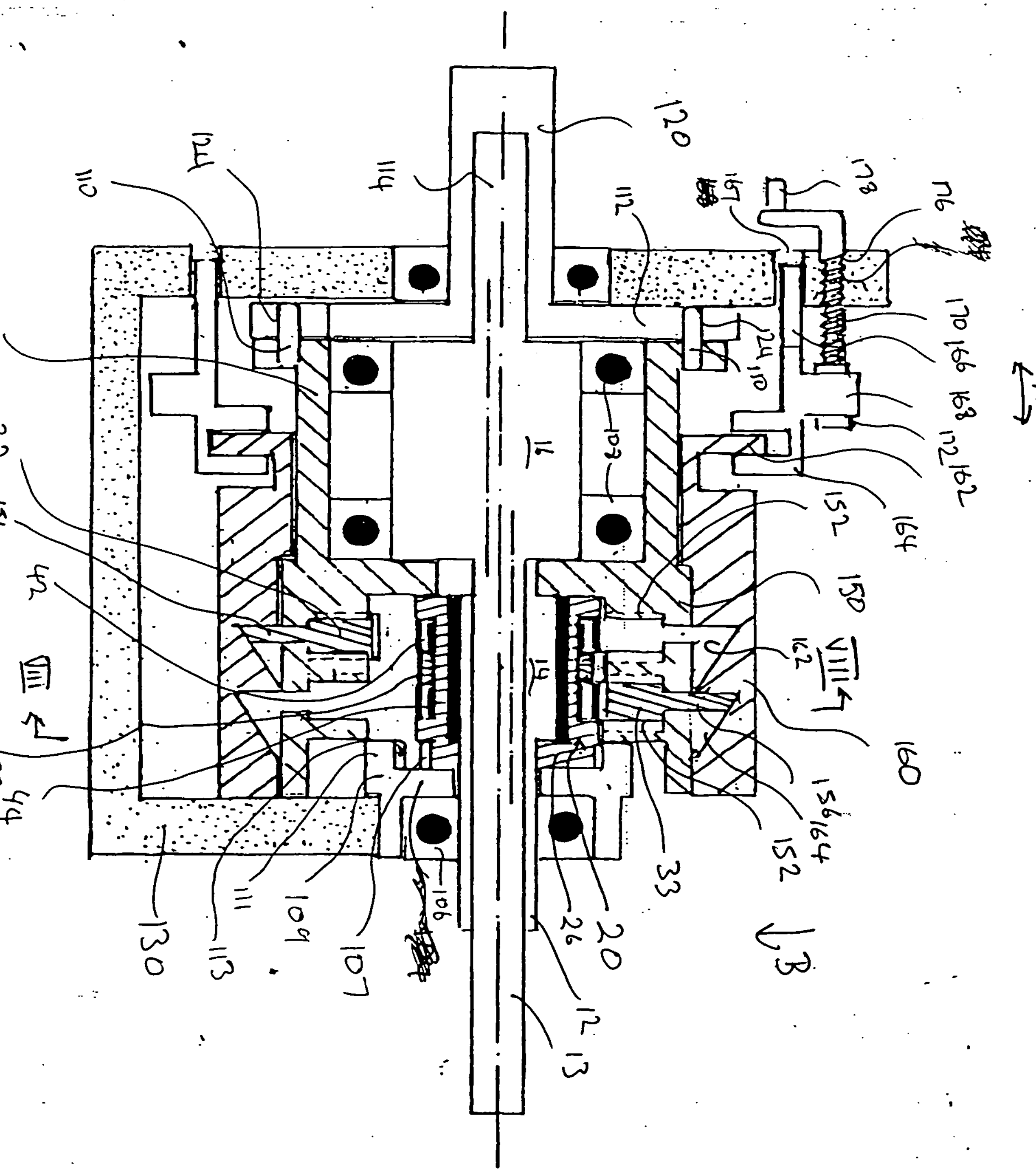


Fig 6



120
114
112
110
124
178
176
170
166
168
172
162
164
152
150
162
VIII
160
156
164
152
33
20
26
12
13
B
A
16
14
107
50
32
158
42
VIII
30
44
113
111
109
107
106
130

Fig 7

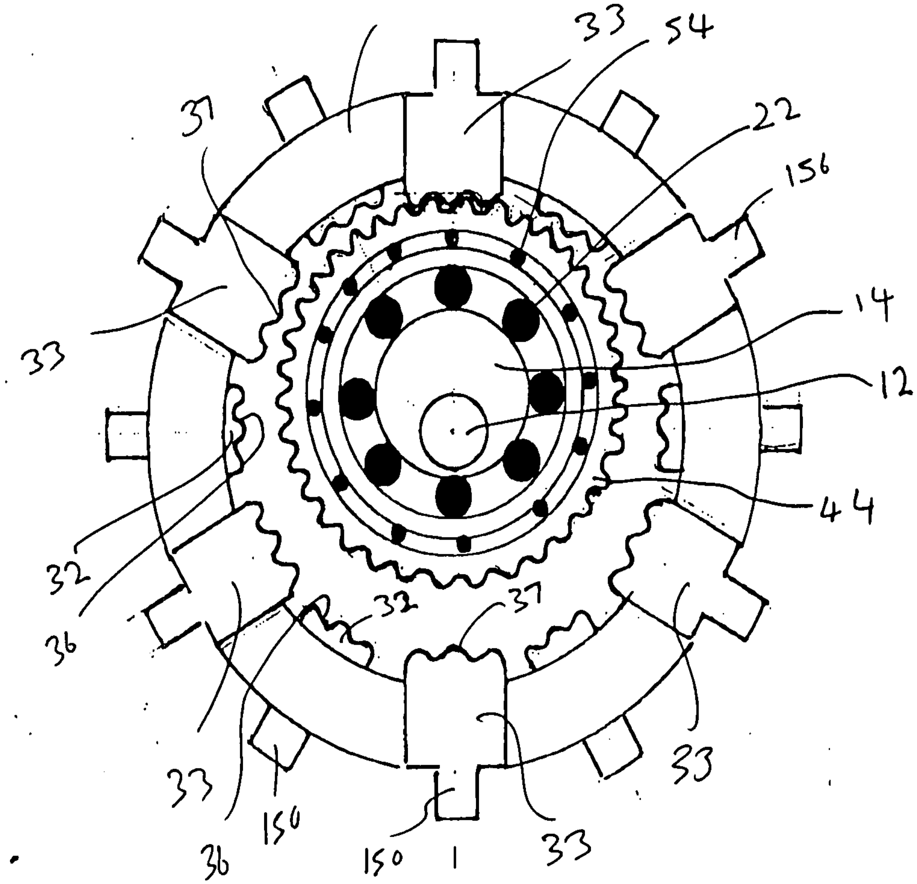


Fig 8

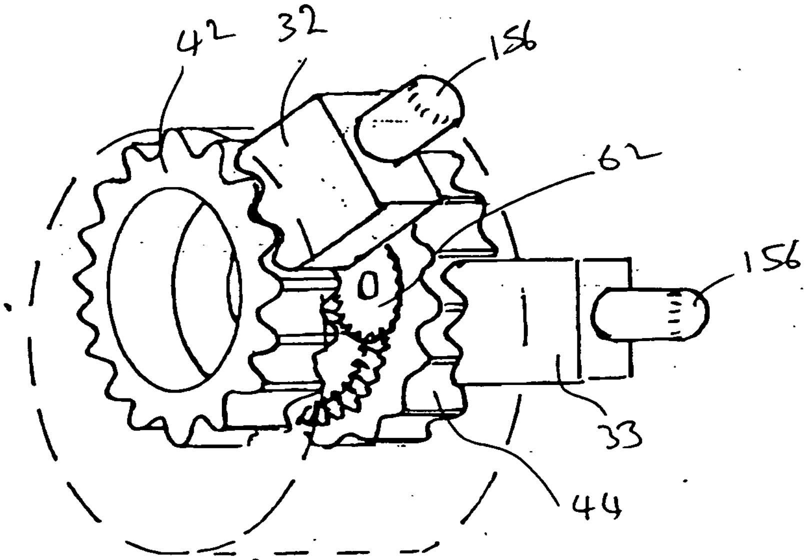


Fig 9

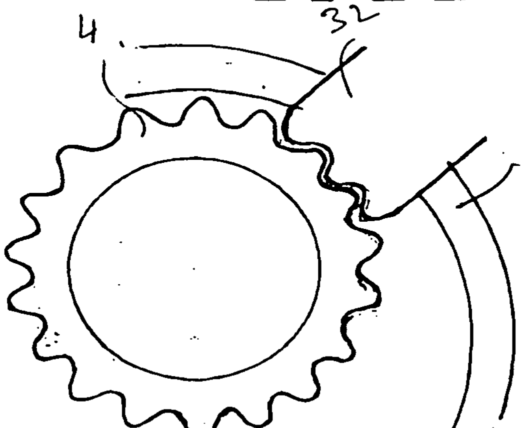


Fig 10A

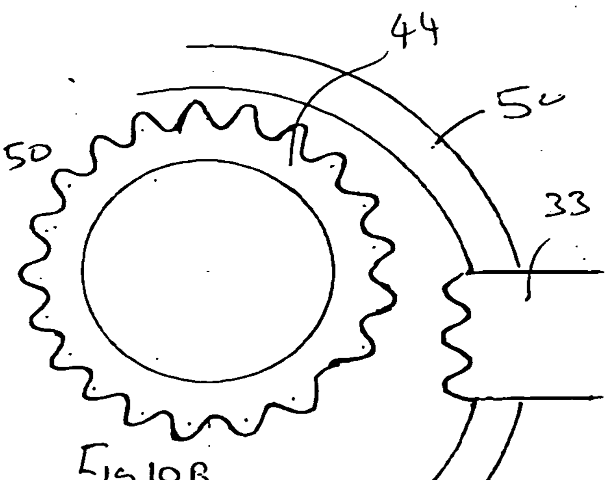


Fig 10B

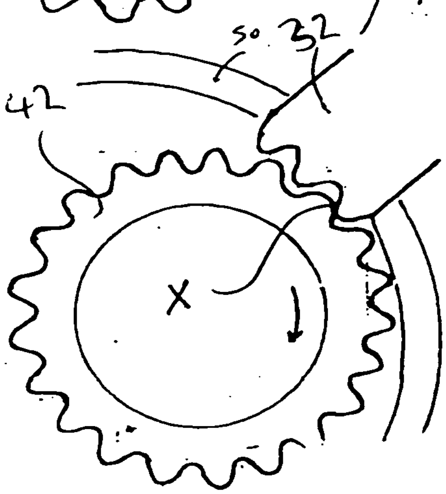


Fig 11A

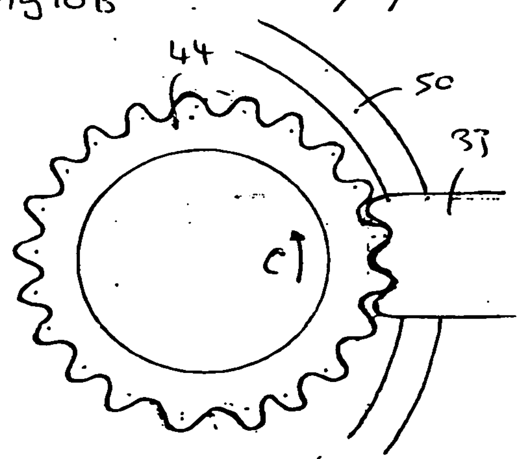


Fig 11B

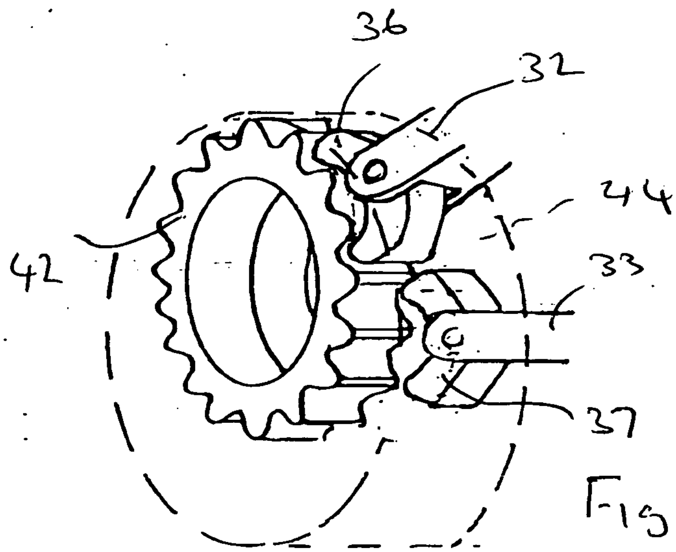


Fig 12

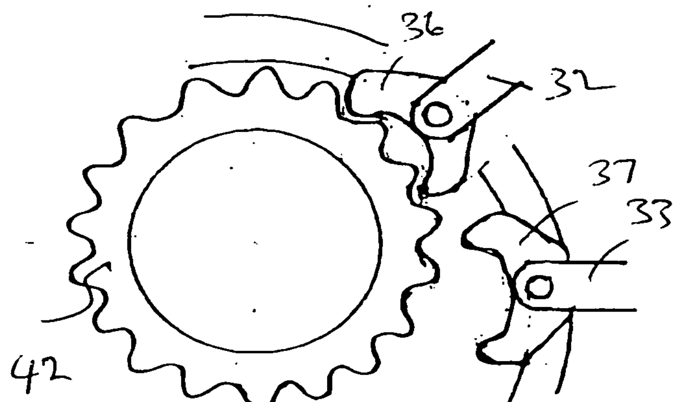


Fig 13

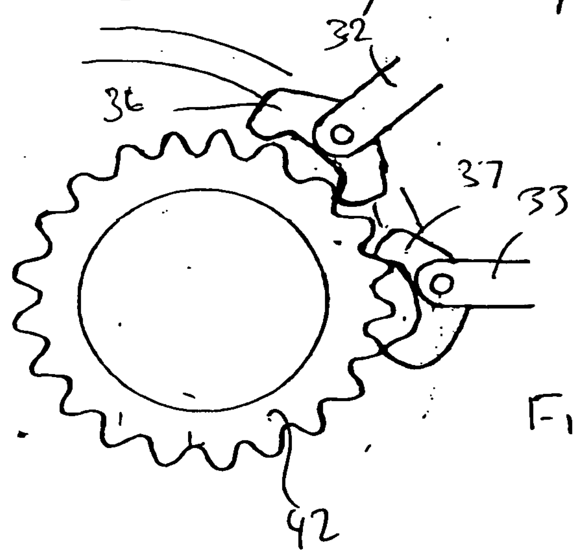


Fig 14

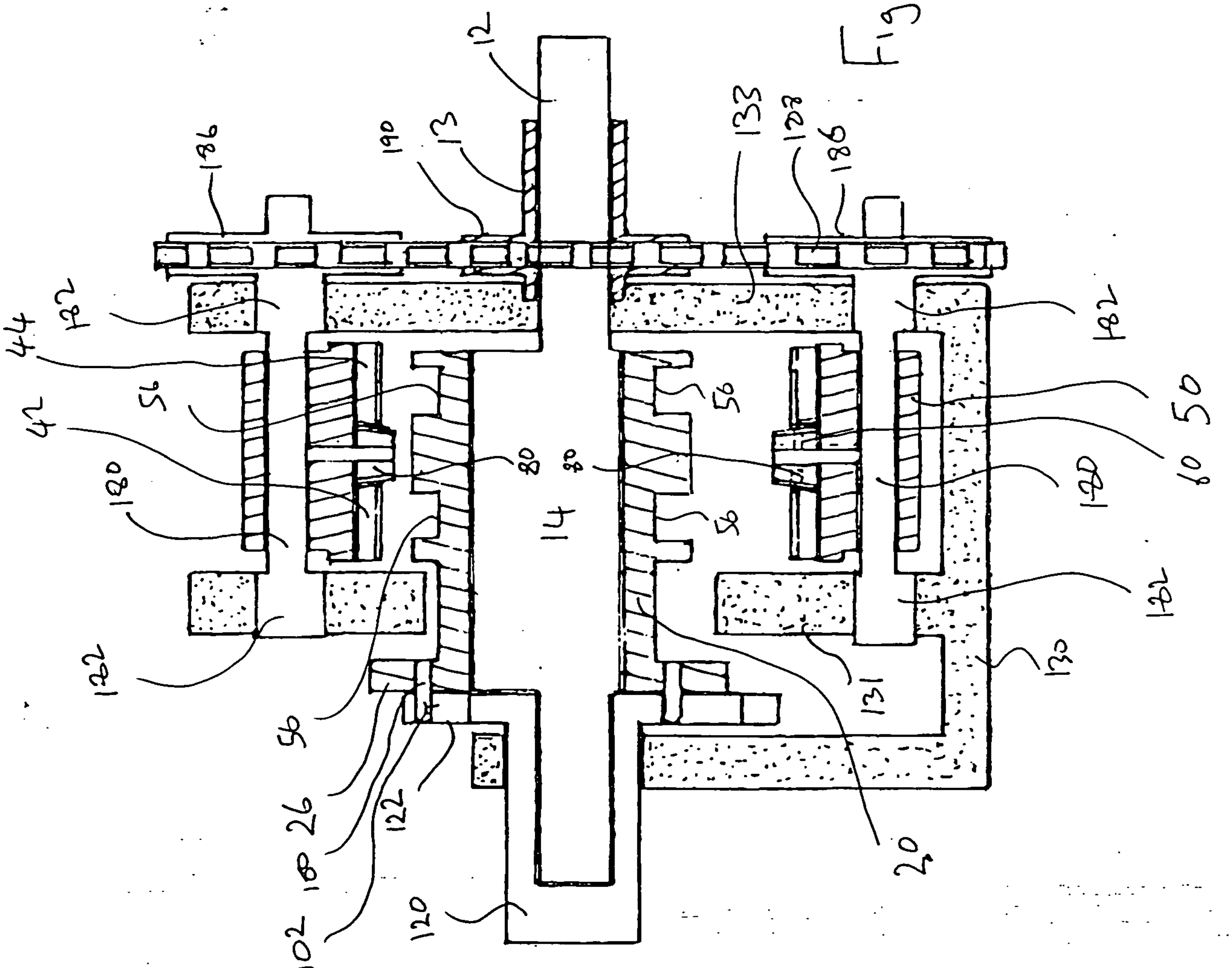


Fig 15